Analysis and Optimization of Interior Noise Based on Transfer Path Analysis

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Abstract. The identification and solution to noise and vibration problems are important ways to continuously improve the comfort in the mechanical field. The car is a more complex mechanical system, and the transfer path of noise and vibration into the car is very complex. It is difficult to identify the path problem by conventional noise and vibration test and analysis measures. Using the transfer path analysis method, the key transfer path of noise and vibration can be identified more quickly. Aiming at the problem of 340–440 Hz and 460–560 Hz acceleration roughness noise in a newly developed car, the OTPA model is built, and the main contribution paths of the problematic frequency band of the car are the right suspension active side tie rod and the right suspension passive side bracket. The scheme of adding vibration absorber is verified, and the acceleration roughness amplitude in this frequency band is reduced by about 3–5 dB (A). The NVH problem can also be solved by increasing the dynamic stiffness of the active side mount bracket with CAE. It is proved that by further improving the dynamic stiffness of the active end mount bracket, the resonance amplitude of the passive end can be reduced, which provides a new idea for similar problems.

Keywords: Noise · OTPA · Transmission path · Contribution analysis

1 Introduction

With the development of science and technology, consumers have higher and higher requirements for the comfort of automotive products, especially the NVH (Noise, Vibration, Harshness) performance of the vehicle, which directly affects the customer's evaluation of the vehicle quality. There are many factors that affect the NVH problem of vehicles. The engine and transmission are the main noise sources. Transfer paths such as mounting system and connecting pipeline will transmit the vibration and noise of the powertrain to the vehicle, thus reducing the quality of the vehicle [1–3].

In the research and development process of a GEELY vehicle, it is found that there are obvious 340–440 Hz and 460–560 Hz acceleration knocking noises in the vehicle, which seriously affect the NVH quality of the vehicle. Since the powertrain is equipped on other vehicles without this problem, it is considered that the biggest cause of the

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problem is the transfer path resonance, which amplifies the noise inside the vehicle, thus leads to this problem. Therefore, the key to solve this problem is to identify the transfer path. In this paper, through the OTPA analysis method, the transfer path of the vehicle is analyzed, and the main contribution path of noise is found. The accuracy of this method is verified by adding vibration absorber, and an effective engineering solution is further found by CAE method.

2 Principle of OTPA Analysis Method

In 1997, P. J.G. Vander Linden et al. and Wim Hendricx et al. introduced the basic principle of this method of quantifying the airborne noise, and elaborated and analyzed the contribution of the body parts which affect the interior noise; In 2016, Guo Shihui and others introduced the transfer path analysis method under working condition load, and introduced the basic principle of TPA (Transfer Path Analysis); In 2019, Jiang Shaowei conducted the research on vehicle interior noise identification based on working condition transfer path analysis method. Through the comparison of research methods in domestic and abroad, the faster test method is operational transfer path analysis (OTPA), which can quickly analyze the contribution of each transfer path to the sound pressure at the measured position, and whether the contribution is caused by large excitation or sensitive path [4–6].

The principle of TPA is to decompose the noise of the response point into the superposition of the different excitation onto each path. The formula is

$$X(\omega) = \sum_{i=1}^{n} H_i(\omega) F_i(\omega) \tag{1}$$

Where $X(\omega)$ is the response and $H_i(\omega)$ is the transfer function of each path, $F_i(\omega)$ for each excitation. For noise response

$$\mathrm{P}(\omega) = \sum_{i=1}^{n} NTF_i(\omega) F_i(\omega) + \sum_{j=1}^{m} NTF_j(\omega) Q_j(\omega) \tag{2}$$

Where $F_i(\omega)$ is the load of each vibration source of the system, $Q_i(\omega)$ is the sound load of each sound source acting on the system, and $NTF_i(\omega)$, $NTF_j(\omega)$ is the transfer function from the excitation point of vibration source and sound source to the response point respectively.

OTPA is based on the response and the transitivity matrix of the response [7–14]. The main theoretical formula is as follows

$$\begin{bmatrix} H_{11} & \cdots & H_{1n} \\ H_{21} & \cdots & H_{2n} \\ \vdots & \vdots & \vdots \\ H_{(m-1)1} & \cdots & H_{(m-1)(\mathrm{n}-1)} \\ H_{m1} & \cdots & H_{m\mathrm{n}} \end{bmatrix} = \begin{bmatrix} X_{11} & \cdots & X_{1m} \\ X_{21} & \cdots & X_{2m} \\ \vdots & \vdots & \vdots \\ X_{(j-1)1} & \cdots & X_{(j-1)(\mathrm{m}-1)} \\ X_{j1} & \cdots & X_{jm} \end{bmatrix}^{-1} \begin{bmatrix} Y_{11} & \cdots & Y_{1n} \\ Y_{21} & \cdots & Y_{2n} \\ \vdots & \vdots & \vdots \\ Y_{(j-1)1} & \cdots & Y_{(j-1)(\mathrm{n}-1)} \\ Y_{j1} & \cdots & Y_{j\mathrm{n}} \end{bmatrix} \tag{3}$$

Where H is the transfer function matrix, X is the input signal matrix (including vibration signal and noise signal), Y is the target point response matrix, m is the number of channels at the excitation point, j is the number of test steps (it is recommended that j is greater than or equal to 3), and n is the number of channels at the response point. After the singular value decomposition of the X matrix, the

$$\begin{bmatrix} X_{11} \\ X_{12} \\ \vdots \\ X_{1(m-1)1} \\ X_{1\mathrm{m}} \end{bmatrix} \begin{bmatrix} H_{11} \\ H_{21} \\ \vdots \\ H_{(m-1)1} \\ H_{m1} \end{bmatrix} = \begin{bmatrix} Y_1 \\ Y_2 \\ \vdots \\ Y_{(m-1)} \\ Y_m \end{bmatrix} \sum\nolimits_{\rightarrow} \ldots Y_{1,syn} \begin{bmatrix} Y_{1,syn} \\ Y_{2,syn} \\ \vdots \\ Y_{(m-1),syn} \\ Y_{m,syn} \end{bmatrix} \sum\nolimits_{\rightarrow} \ldots Y_{syn} \tag{4}$$

The singular value decomposition of the X matrix also realizes the main component decomposition. The smaller part of the eigenvalues need to be discarded in the calculation.

3 OTPA Transmission Path Identification

The main parameters of the vehicle studied in this paper are shown in Table 1. The main problem frequency bands of the car in the partial load acceleration condition are 340–440 Hz and 460–560 Hz. The spectrum diagram of interior noise is shown in Fig. 1. As can be seen from Fig. 1, the energy of these two bands is very prominent during the whole operation condition, with which can be judged as two resonance bands. Because the masking effect of road noise, wind noise and other noises are relatively small at low vehicle speed which resulting the noise of these two bands are very obvious in the vehicle carbine, and consequently sound quality of the vehicle is seriously affected. In the total cost of power, the vibration signals of the two resonance bands are not obvious, so it can be judged that there is resonance of parts in the transmission path, which causes problems in the car.

Table 1. Vehicle and engine parameters

Vehicle	Engine					
Weight	Displacement	Air influence	Injection method	Rated power	Rated speed	Maximum torque
1670 kg	1.969 L	Turbocharged	Direct injection	175 kW	5500 r/min	350 Nm

The structure transfer path of the car mainly includes the suspension system, heating water pipe, air conditioning pipe, transmission half shaft, shift cable, exhaust hook, etc. the radiated noise is mainly transmitted into the car through the front wall of the car body. Therefore, OTPA method can be used to analyze and sort the possible structure

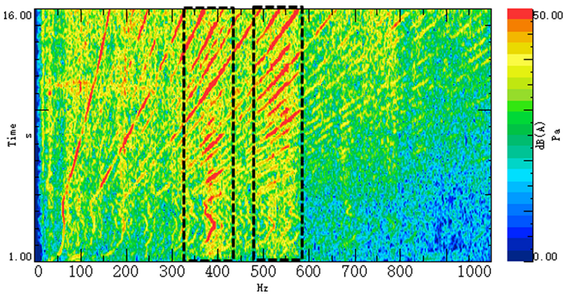

Fig. 1. Interior noise spectrum under acceleration condition

contribution path and air contribution path, and quickly identify the main contribution path. In OTPA analysis, it is necessary to fully understand all the transmission paths of the whole vehicle, and the layout of the measurement points of the required path is highly required. If the selection is not reasonable, the transfer function error will be relatively large, which will affect the accuracy of the noise analysis of the powertrain in the vehicle. The transfer function is calculated under different working conditions, so there are high requirements for the selection of test conditions. Therefore, the sensors should be arranged carefully and the operating conditions should not be similar to each other [14, 15].

This paper mainly considers four mounting points, left and right axle, air conditioning pipeline and heating pipeline in the structural path analysis by totally dampening the intake and decoupling the exhaust hook. The acceleration sensor is used to obtain the vibration data of each structural path as the excitation input of the structural path. The air path mainly considers the influence of cabin on the interior noise. The microphone measurement points are in the near field of engine, transmission and generator transformer. The data of near field noise is used as the excitation input of air path. Therefore, the specific digital analysis model and sensor test points are shown in Fig. 2 and Fig. 3.

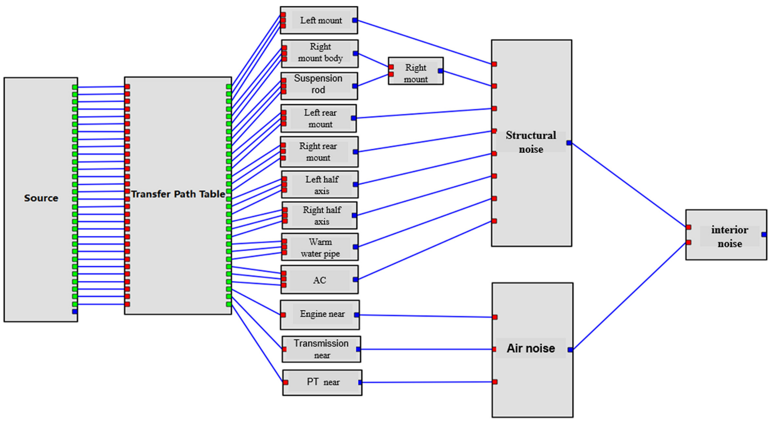

Fig. 2. OTPA calculation model

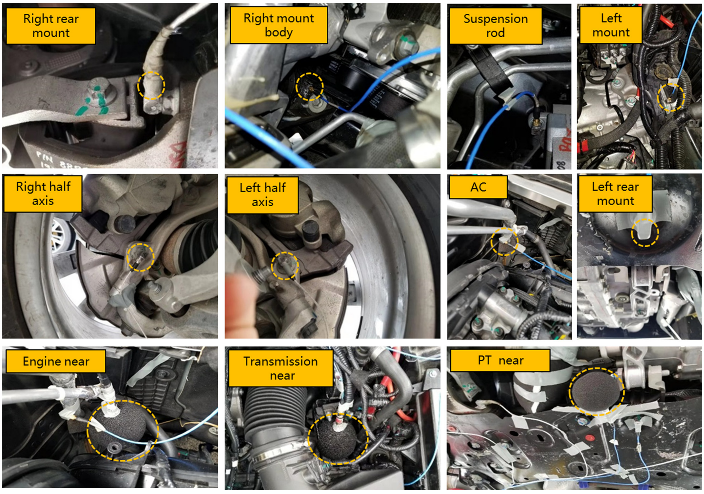

Fig. 3. Sensor pointing photos

In order to get a more accurate transfer function, not only the main path measurement points should not be lost and repeated, but also the number of working steps should not be less than 3 times of the number of channels. In this paper, the working condition data of 30% throttle opening is used to analyze the problem, and the rest of the working condition data is used to solve the transfer function. As shown in Fig. 4, the spectrum comparison between the test results of 30% throttle opening and the calculation results of vehicle interior noise shows that the calculation results of the problem frequency band are close to the measured results, which indicates that the model has included the main contribution path of the problem and ensures the accuracy of path contribution analysis.

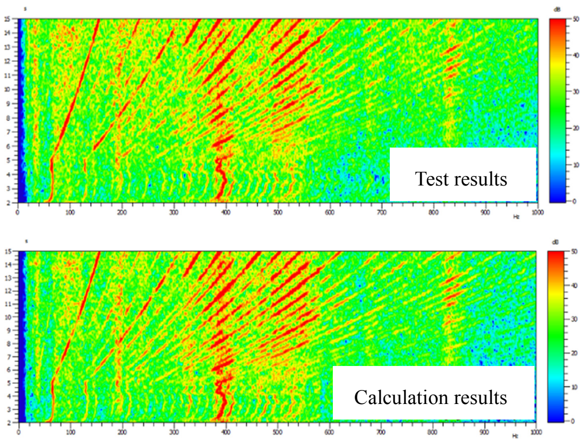

Fig. 4. In-vehicle noise test results and OTPA model calculation results

Figure 5 shows the analysis results of each path contribution of vehicle interior roughness sound. It can be found that the paths that have the greatest impact on vehicle interior roughness sound are the right suspension rod and the right suspension passive side bracket. In order to reduce the rough sound and improve the interior sound quality, these two main paths need to be optimized and verified.

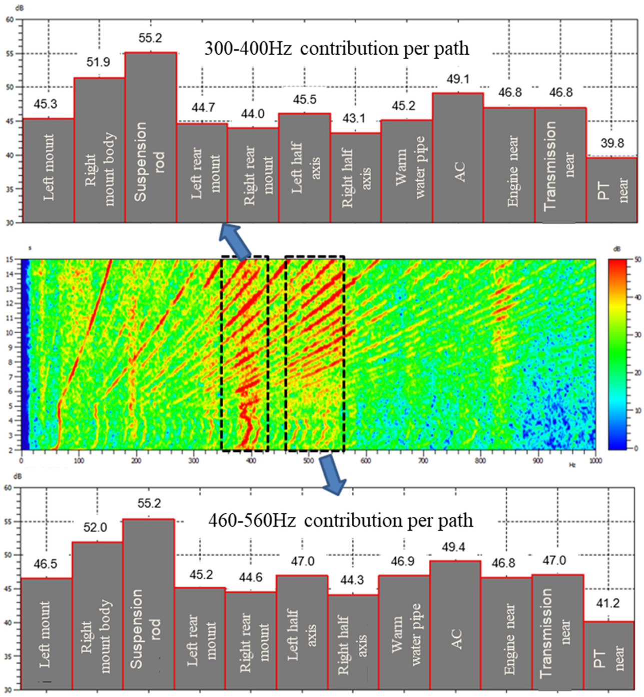

Fig. 5. Pass path contribution analysis results

Figure 6 and Fig. 7 show the vibration spectrum changes of the right suspension rod and the passive side bracket with vibration absorber added respectively. It can be concluded from the comparative effect of the frequency color charts in Fig. 6 and Fig. 7 that after adding the vibration absorber, the vibration of the right suspension rod and the passive side bracket decreases significantly, and the vibration peaks at the characteristic frequency bands of 340–440 Hz and 460–560 Hz basically disappear. It can be seen from Fig. 8 and Fig. 9 that after adding vibration absorbers to the right suspension rod and the right suspension passive side bracket, the amplitude of 340–440 Hz and 460–560 Hz acceleration roughness sound in the vehicle decreases by about 3–5 dB (A), the subjective evaluation of the interior roughness sound basically disappears, and the sound quality of the interior acceleration condition is significantly improved. It also proves that the path which has the greatest impact on the noise in this frequency.

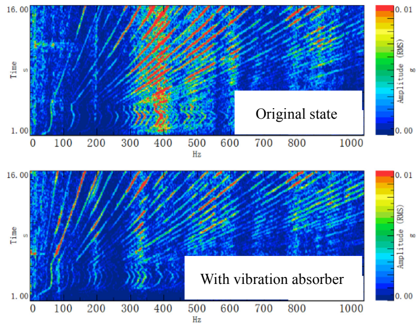

Fig. 6. Vibration comparison of right suspension rod with vibration absorber

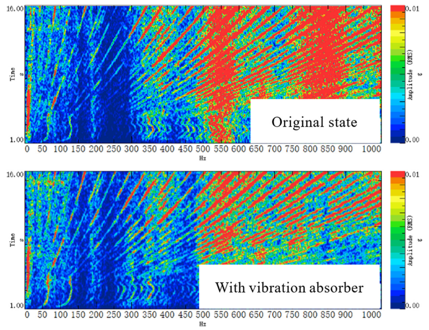

Fig. 7. Vibration change of right mount passive side bracket with vibration absorber added

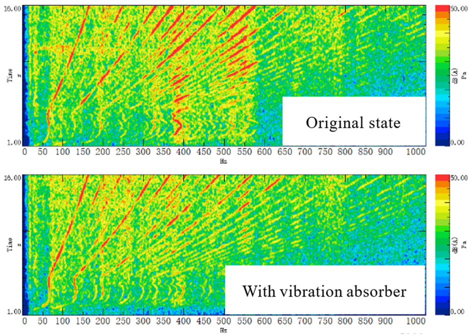

Fig. 8. Comparison of interior noise of vehicle with vibration absorber on passive side of right mount

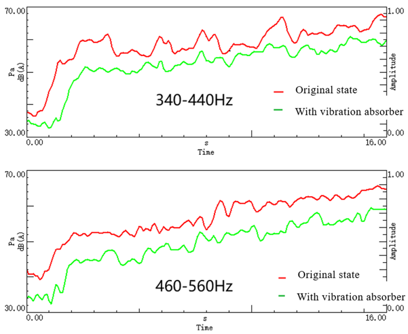

Fig. 9. Comparison of noise frequency slice in vehicle with vibration absorber added to right mount

4 CAE Optimization Analysis of the Delivery Path

It can be seen from the previous text that the right suspension rod and the right suspension passive side bracket are the main paths for the noise transmission of the vehicle. Obviously, the resonance of the two brackets enlarges the excitation of the powertrain, resulting in the noise problem in the vehicle. Although this problem can be solved by adding vibration absorbers, it involves the layout space and will increase the cost. It is obviously very difficult to directly change the suspension passive side bracket or even the body sub frame.

According to the formula of transmission rate of mount, the influence of transmission path can be reduced by optimizing the active side and passive side of mount. When the structural characteristics of the mount remain unchanged, the transmissibility is only related to the frequency ratio and damping ratio. When the mount attenuates the excitation (without resonance), the amplitude of the force transmitted to the foundation is proportional to the amplitude of the excitation force. Reducing the excitation force of the active side mount can also reduce the excitation transmitted to the passive side of the car body. The formula of transmissibility is as follows:

$$T = \left|\frac{F_a}{F}\right| = \sqrt{\frac{1 + (2\xi r)^2}{(1 - r^2)^2 + (2\xi r)^2}} \tag{5}$$

Where T is the transmissivity, F_α is the amplitude of the force transmitted to the foundation, F is the amplitude of the exciting force, r is the ratio of the exciting frequency to the natural frequency, and ξ is the viscous damping ratio [15].

Therefore, the author considers to optimize and improve the active side of the right mount. By reducing the dynamic flexibility of the active side of the right mount, the vibration excitation force transmitted from the powertrain to the vehicle body is reduced, and the risk of noise transmission to the vehicle is reduced. Therefore, the CAE simulation of powertrain is established, as shown in Fig. 10. The simulation model of CAE powertrain system includes cylinder head, cylinder block, crankcase, cylinder head cover, mounting bracket, oil pan, right mounting bracket, engine cooling bracket and other parts system. The engine moving parts system is simulated by concentrated mass and divided by two-order tetrahedron, with a total of 700,000 units and 1.2 million nodes. The boundary condition is: in the free state of the whole movable assembly, unit load is applied to the center point of the right suspension active end bracket for calculation, and the origin dynamic stiffness of the suspension active end bracket in the assembly state of the movable assembly is finally obtained.

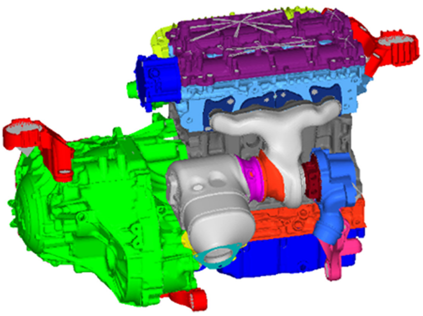

Fig. 10. Build the powertrain simulation model

The analysis results are shown in Fig. 11. The blue line is the original dynamic flexibility of the active end bracket, and the green line is the optimized dynamic flexibility of the active end bracket. It can be seen from the figure that the Z direction is much larger than the empirical value of 1e−4 mm/N, so it is necessary to reduce the dynamic flexibility and improve the dynamic stiffness of the active side bracket. The dynamic stiffness of the active side bracket is improved by adding a bolt connection. The results of dynamic flexibility analysis after adding bolts are shown in the green line of Fig. 11. The dynamic flexibility of the z-direction active side bracket is reduced by about 72% (the peak value is reduced from 6.7e−04 mm/N to 1.85e−04 mm/N), and the first-order modal frequency of the active side bracket is increased from 820 Hz to 993 Hz. At the same time, the dynamic flexibility decreases about 60% at 340–440 Hz and 64% at 460–560 Hz, which reduces the vibration response of the active side of the mount and controls the vibration transmission to the passive side of the mount. After optimization, the interior noise is reduced by about 2 dB(A) in the range of 340–560 Hz, which basically solves the problem. Therefore, although the problem of the vehicle is caused by the resonance of the right suspension rod and the right suspension passive side bracket, in the actual project, when various restrictions cannot be changed, the problem can be solved by greatly improving the dynamic stiffness of the active end bracket, reducing the engine transmission excitation, and reducing the vibration transmission rate. This idea has important reference significance for the optimization of transfer path of practical engineering problems.

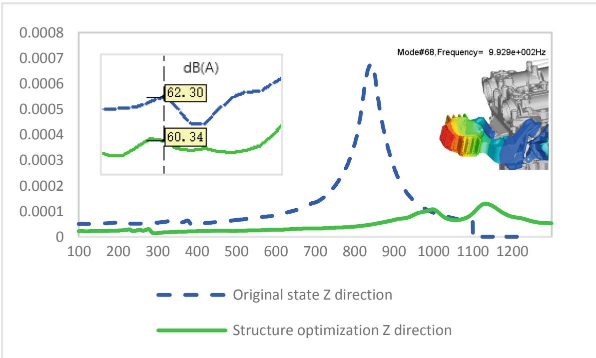

Fig. 11. Test results of structure optimization analysis of right suspension active side bracket

5 Conclusion

In this paper, based on the main noise frequency band of a car powertrain, the digital analysis of transmission path is carried out. Using the test and analysis method of working condition transmission path, the contribution order of the noise of the car in 340–440 Hz and 460–560 Hz is quickly found, and the main contribution path of each noise area is obtained. The main paths are the right suspension rod and the right suspension passive side bracket. The main conclusions are as follows.

(1) OTPA transmission path analysis method plays an important role in the vibration and noise analysis of the whole vehicle. Through this method, the main noise contribution paths of the powertrain can be quickly identified, and the contribution sizes can be sorted, so as to provide the basis for solving the problem quickly;
(2) For the design of the mounting system, not only the vibration isolation rate of the system should be considered, but also the design of the mounting bracket is very important. It is necessary to control the resonance frequency and dynamic stiffness of the active and passive end bracket of the mounting bracket, so as to reduce the transmission of vibration and the noise inside the vehicle as much as possible;

(3) In practical engineering, the resonance of the passive end of the mounting bracket can be controlled by improving the dynamic stiffness of the active end, so as to reduce the impact on the interior noise, which also has important reference significance for the optimization of the transmission of other paths.

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